Numerical Studies on Geometric Features of Microchannel Heat Sink with Pin Fin Structure

Jin Zhao, Shanbo Huang, Liang Gong*
* Corresponding author: Tel.: +86 (0)532-86981767; Fax: ++86 (0)532-86981822; Email: lgong@upc.edu.cn
Department of Energy & Power Engineering, China University of Petroleum (Huadong), Qingdao, China

Abstract The manifold microchannel (MMC) heat sink is an effective method of electronic device cooling featured by the advantages such as low thermal resistance, compact structure, low flow rate and uniform temperature distribution along the flow direction. To improve the performance of MMC, numerical studies on geometry features of pin fin microchannel were carried out and the characteristics of flowing and heat transfer in two types of MMC structures which are respectively optimized through porosity with different pin-fin distribution and pin-fin located angle were investigated. Numerical results indicated that there are apparent influences of both above geometry features on the characteristics of flowing and heat transfer in MMC. Based on cooling performance factor, an optimum value of porosity with specific pin-fin distribution was obtained. Moreover, better effect on heat transfer could be achieved at a 30 degree around of rectangular pin fin located angle in our work.

Key words: pin-fin microchannel, geometric features, numerical simulation, thermal performance

1. Introduction

Along with the power density of integrated circuits continuously and rapid rising, technical problem on thermal management becomes the most prominent obstacle to scaling down the size of modern electronics [1]. In this situation, the research issued on the flow and heat transfer characteristics under micro scale was initially applied in microelectronics field [2]. Due to its high packing density, small mass and volume as well as larger area to volume ratio, micro-channel heat sink cooling technology has been gradually focused on for cooling high heat flux components and become a new hotspot of cooling technology research [3, 4]. One of the various solutions proposed is to enhance heat transfer by adding pin fin structure in micro-channel heat sink, also named MMC (Manifold Micro-Channel type). Recent studies have shown that the latter provides a larger heat transfer area per volume in addition to higher heat transfer coefficients due to boundary layer disruption and other mixing effects [5, 6]. Numerous analytical, experimental and numerical studies have been conducted to investigate on pin fin micro-channel heat sinks especially in the area of geometric optimization. One major structure optimization attempts to study the micro-channel pin fin heat sink with different fin geometries and cooling fluids. Ricci et al [7] experimentally investigated the pin fin heat sink with fins in different shapes (circular, square, triangular and rhomboidal) arranged in line with constant heat flux boundary condition. Hasan [8] numerically investigated the micro-channel pin fin heat sink with three fin geometries and two types of nanofluids in addition to the pure water. Kosar et al [9-13] experimentally investigated the pressure drop and friction factors in four micro heat sink configurations (circular, rectangular, diamond shaped, hydrofoil shaped and cone shaped geometries) cooled with water and R-123 fluid and proposed a modified friction factor correlation. Additional optimization study has focused on the different fin arrangement of micro-channel pin fin heat sinks. Liu et al [14] experimentally studied the performance of pressure drop and heat transfer characteristics in micro square pin fin heat sink with staggered arrangement, using deionized water as working fluid. Shafeie [15] presented a numerical study of laminar forced convection in heat sinks with micro pin-fin structure. Among different layouts for the pin fins such as oblique and staggered distributions, the case with the oblique orientation of short pins had the highest heat removal for a given pumping power.

In general, most of the previous studies of micro-channel pin fin heat sink focused on optimizing the fin geometrical shape, cooling fluids and the fin arrangement. However, for achieving the optimized structure of rectangular micro-channel pin fin heat sink, it still needs to further study on the characteristics of flowing and heat transfer in MMC heat sink with pin-fin layout. Therefore, based on the idea of porous medium, numerical studies on geometry features of micro-channel pin fin heat sink were carried out in this paper to improve the performance of MMC. The geometry feature factors of porosity and pin fin rotated angel were selected to optimize the layout of MMC heat sink. The characteristics of flowing and heat transfer in two types of MMC structures were investigated.

2. Computational Model

2.1 Physical Models

Based on the concept of porous media, for MMC heat sink, the parts of pin fins and channels can be seen as the skeletal portion and the pores of foams structure respectively. Therefore, the volume of fluid flowing region on the entirely structure is called porosity. Considering the layout, the MMC heat sink with pin-fin is shown in Fig. 1, where W and H are the width and height interval distance between pin fins respectively.

For optimizing the structure of MMC heat sink, the porosity and the rotated angel of pin fin shown in Fig.2 were considered as two optimal parameters. The corresponding models of micro-channel heat sinks with pin-fins are named model 1 and model 2, as shown in Fig3, 4. Model 1 was designed based on the porosity from 0.56 to 0.89. The details of structure parameters are provided in table 1. In the model 2, the rotated angels of θ are from 0 to 45 degree and the porosity is set as 0.75.

Aiming to investigate the flow and heat transfer characteristics of rectangular pin fin heat sink under different pin fin rotated angel and porosity, the same fluid flow condition was set in both model 1 and model 2. Moreover, metal foams with the porosity of 0.9 were filled in the inlet collector for uniform mass flux distribution. It is noticeable of that both channels diameters at the inlet and outlet of all the heat sinks are 2 mm. The width of each single channel inside is 1 mm. The working fluid used in all conditions are water-liquid and it's detail thermodynamic properties are as follows: ρ=998.2 kg/m^3, c_p=4182 J/(kg·K), λ=0.6 W/(m·K), η=0.001003 kg/(m·s), where ρ, c_p, λ, η means density, specific heat, thermal conductivity, viscosity. In addition, the physical dimensions of all microchannel heat sinks are unified; the length, width and height are 22 mm, 19 mm and 6mm respectively.

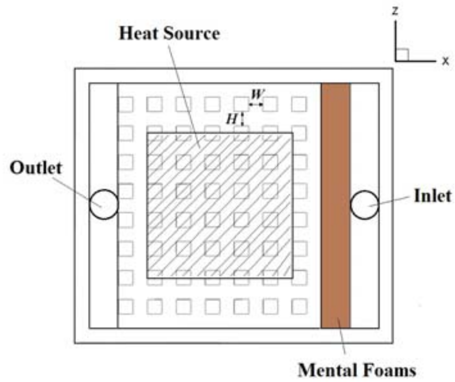


Fig. 1 Structure diagram inside the MMC heat sink

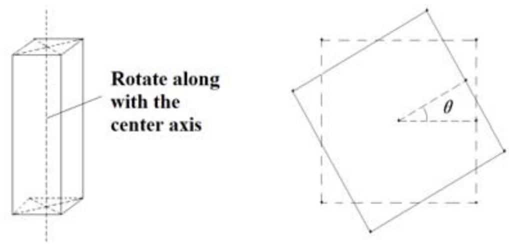
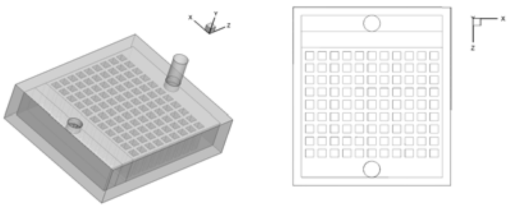


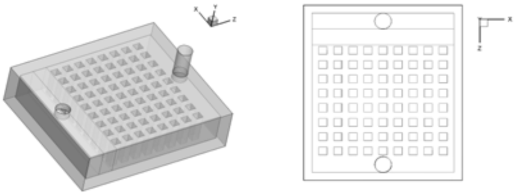
Fig.2 The definition of the pin fin rotated degree θ

Table 1 Porosity of microchannel heat sinks

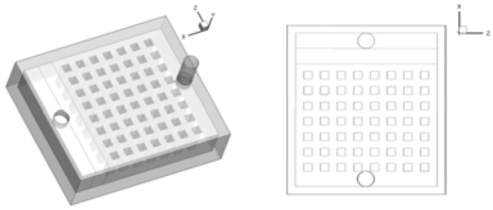
Feature size		Porosity
W/ mm	H/ mm	
0.5	0.5	0.56
0.8	0.8	0.69
1	1	0.75
1.2	1.2	0.79
1.5	1.5	0.84
2	2	0.89



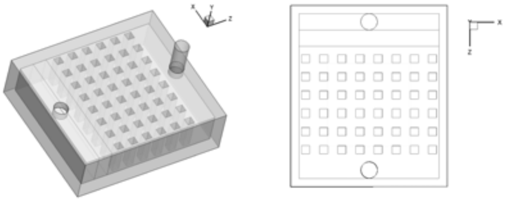
Model 1(a) ε=0.56 (W=0.5 mm, H=0.5 mm)



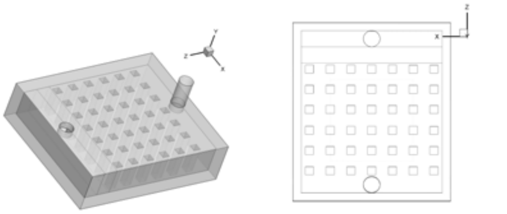
Model 1(b) ε=0.69 (W=0.8 mm, H=0.8 mm)



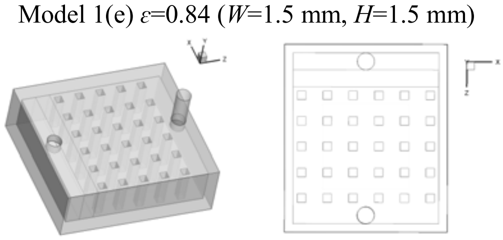
Model 1(c) ε=0.75 (W=1.0 mm, H=1.0 mm)



Model 1(d) ε=0.79 (W=1.2 mm, H=1.2 mm)

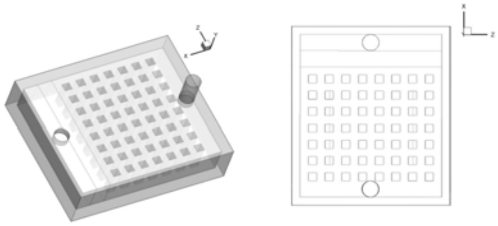


Model 1(e) ε=0.84 (W=1.5 mm, H=1.5 mm)

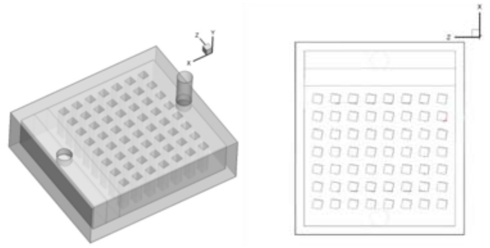


Model 1(f) ε=0.89 (W=2.0 mm, H=2.0 mm)

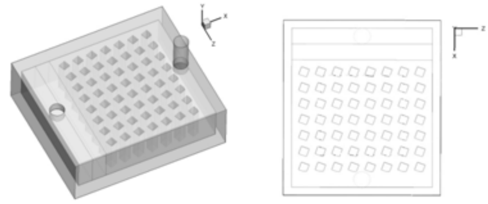
Fig. 3 Physical structures of model 1: optimized by porosity with specific pin-fin distribution



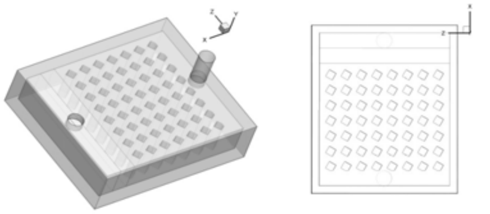
Model 2(a) θ=0^{o}



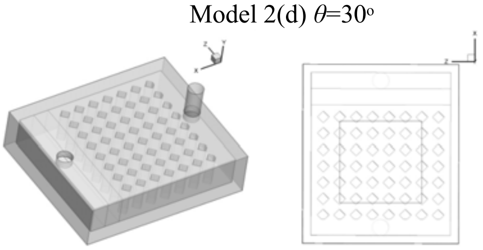
Model 2(b) θ=10^{o}



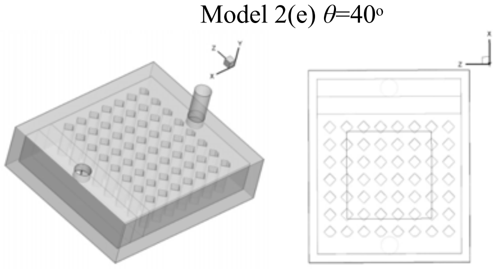
Model 2(c) θ=20^{o}



Model 2(d) θ=30^{o}



Model 2(e) θ=40^{o}



Model 2(f) θ=45^{o}

Fig. 4 Physical model of model 2: optimized by pin fin rotated angel

2.2 Mathematical Model

2.2.1 Assumptions

To make the problem more reasonable and simpler, several assumptions were employed:
1) Steady, laminar flow;
2) No thermo-physical property variation with temperature;
3) Incompressible Newtonian fluid;
4) No velocity-slip boundary;
5) Continuous temperature and heat flux.

2.2.2 Governing Equations

The steady-state conservation equations for mass, momentum, and energy in fluid are given by

$$\nabla \cdot (\rho \mathbf{u}) = 0 \tag{1}$$

$$(\mathbf{u} \cdot \nabla) \rho \mathbf{u} = -\nabla p + \mu \nabla^2 \mathbf{u} \tag{2}$$

$$\mathbf{u} \cdot \nabla T = \frac{k \nabla^2 T}{\rho c_p} \tag{3}$$

Energy equation in the solid domain is given by

$$\nabla^2 T = 0 \tag{4}$$

2.2.3 Boundary Conditions

In present work, some unified settings were adopted to all the four microchannel heat sinks. For boundary conditions, velocity inlet and fully developed outlet were set up and all the solid walls were treated as adiabatic. On the bottoms of each heat sink, a constant heat flux 10^6 W/m^2 was assumed in the central area. Working fluid of 300 K with constant physical properties at different inlet velocity of 0.48 m/s, 0.72 m/s, 0.96 m/s, 1.2 m/s, 1.44 m/s were supposed to flow into each inlet pipe, and local unilateralization was idealized as outlet condition.

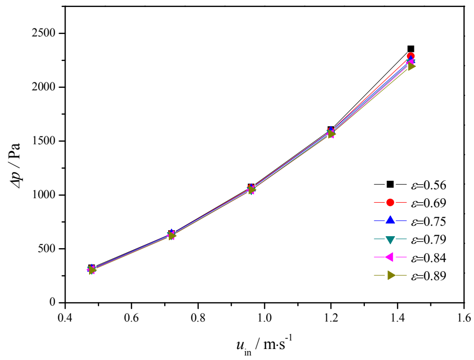
2.2.4 Grid Independency

In consideration of improving convergence speed and computing accuracy, the hexahedral mapped mesh was chosen for our simulations. Several mesh selected with grid numbers from 338101 to 2601377 were employed for checking the mesh independency of numerical solution. By studying, the differences of pressure drop and *Nusselt* Number between 1.4 million mesh and 2.6 million mesh are about 0.4% and 0.3% respectively. As a result, the 1.4 million mesh was the best choice to satisfy efficiency and accuracy at the same time and will be used to all the models in this paper.

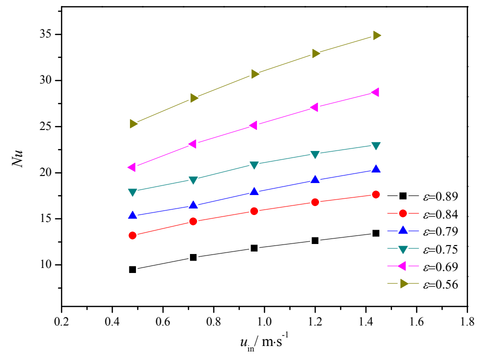
3. Results & Discussion

3.1 Effect of pin fin porosity on cooling capacity

Fig. 5 shows the *Nusselt* Number and pressure drop with the function of pin fin porosity and inlet velocity. It can be observed that both pressure drop and *Nusselt* number increase gradually with the increasing of inlet velocity. From Fig. 5(a), it can be seen that the pin fin porosity is ineffective for flow resistance with lower inlet velocity. But with the increasing of inlet velocity, the effect emerges gradually. The pressure drop is increase almost 15% when the pin fin porosity decreases from 0.89 to 0.56, which results by the decreasing of pin fin porosity can not only reduce the flowing area but also strengthen the fluid disturbance, thus resulting in the increasing of flow resistance. Moreover, with the decreasing of pin fin porosity, the heat transfer is enhanced greatly, as shown in Fig. 5(b). The *Nusselt* number increases almost 3 times, with inlet velocity of 1.44 m/s, when the pin fin porosity decreases from 0.89 to 0.56, which results by that both of the heat transfer area and the fluid disturbance are increased with the amount of pin fin increasing. At the same time, the heat transfer enhancement also results in the cooling capacity of heat sink enhancement, as shown in Fig. 6. From this figure, it can be seen that the maximum temperature of chip surface can be reduced to 321 K from 338 K, with the inlet velocity of 1.44 m/s, when the pin fin porosity decrease from 0.89 to 0.56. Even for the inlet velocity of 0.48 m/s, the T_{max} can be reduced to 332 K from 354 K. Fig. 7 visually shows the positive effect.



(a) Pressure drop



(b) Average *Nusselt* number

Fig. 5 The flow resistance and heat transfer of model 1

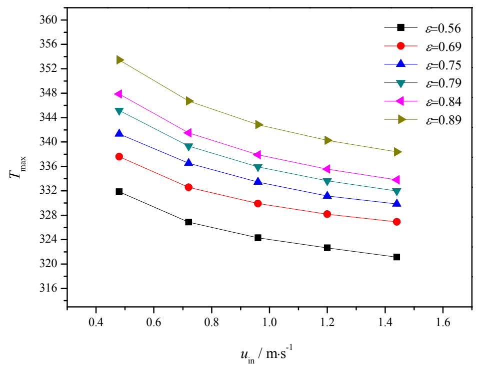
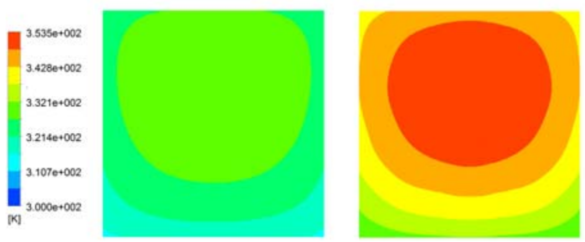


Fig. 6 The maximum temperature of chip surface



(a) ε=0.56 (b) ε=0.89

Fig. 7 Temperature distribution on the surface of heat sources with u_{in}=0.48 m/s

Although the smaller pin fin porosity can enhance heat transfer, it does not mean that the small porosity is always positive for thermal performance. Therefore, a standard criterion which comprehensively accesses the flowing and heat transfer capability on microchannel heat sink is needed, a thermal performance factor [9] was adopted in our work, defined as Eq. 5, which can evaluate the heat transfer efficiency under the same pumping power.

$$pf = \left(\frac{Nu}{Nu^0}\right) \Big/ \left(\frac{\Delta p}{\Delta p^0}\right)^{\frac{1}{3}} \tag{5}$$

Here, Nu, Nu^0 and Δp, Δp^0 represent *Nusselt* number and pressure drop of comparison model and standard model with ε=1, respectively.

Thermal performance factor with different pin fin porosity is shown in Fig. 8. It can be seen that pin fins are positive on heat transfer efficiency as a whole. But it is not consistent, there exists an optimum. From Fig 8, the optimum of pin fin porosity is about ε=1 for the simulation cases. Moreover, Fig. 8 also shows the heat transfer efficiency of the mode with optimal porosity larger than the one of the model without pin fin over 2 times, which indicates manifold microchannel holds the feature of lower flow resistance and higher thermal performance.

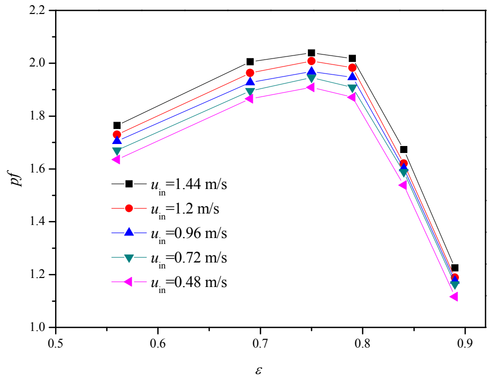
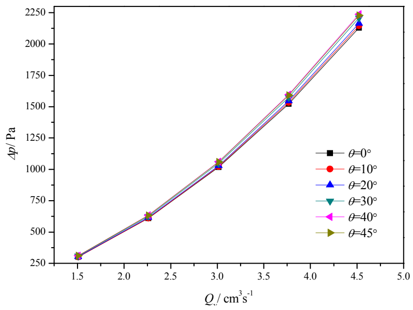


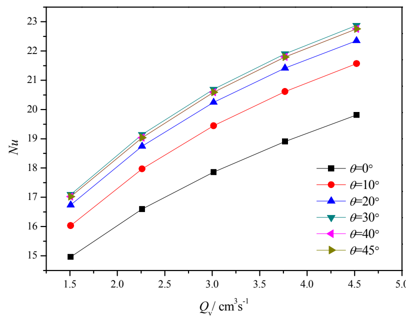
Fig. 8 Thermal performance factor

3.2 Flow and heat transfer analysis on rectangular manifold located angel

For enhancing heat transfer deeply, the pin fin rotated angel, as shown in Fig.3, were numerically studied. Fig. 9 shows the pressure drop and the *Nusselt* number with the pin fin rotated angel from 0° to 45°under different inlet velocity. It can be observed that the pin fin rotated angle is ineffective on flow resistance but greatly effective on heat transfer. As shown in Fig. 9(b), the *Nusselt* number increases gradually with the increasing of pin-fin located angel. Fig. 10 visually shows the positive effect. That is because that the boundary of pin fins is matched with streamline gradually with the increasing of θ, which results in the increasing of convection heat transfer around pin fins since the vortex in region between two adjacent pin fins is disappeared gradually, as the case of θ=0° and θ=30°shown in Fig. 11. However the *Nusselt* number begins to decrease gradually when the pin-fin rotated angel is larger than 30°, as shown in Fig. 9, which is because the boundary of pin fin are not matched with streamline again, as shown in Fig. 11(c), the vortex reappears, resulting in the decreasing of convection heat transfer around pin fins. So, this characteristic indicates that the heat transfer can be enhanced by adjusting the pin fin rotated angle and there exists an optimal pin fin rotated angle. Fig. 12 also indicates the feature clearly. It can be seen that the cooling capacity of MMC heat sink can be enhanced by rotating pin fins. When the pin fin rotated angle equaling to 30 degree, the maximum temperature of chip surface can reach to the minimum value, which also proves the existence of optimal pin fin rotated angle. Fig. 13 visually shows the characteristics. From Fig. 14, it can be shown, for the simulation cases, the optimal pin fin rotated angle is about 30°, and the heat transfer efficiency can increase over 13% with the optimal angle.

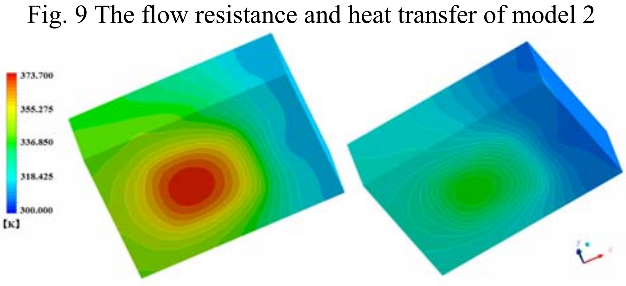


(a) Pressure drop



(b) Average *Nusselt* number

Fig. 9 The flow resistance and heat transfer of model 2



(a) θ=0° (b) θ=30°

Fig. 10 Temperature distribution under u_{in}=0.48 m/s

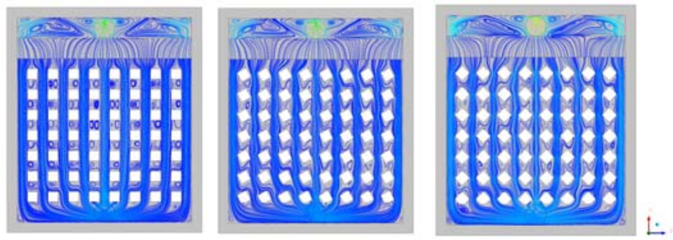

(a) θ=0° (b) θ=30° (c) θ=45°

Fig. 10 Velocity streamline under u_{in}=0.72 m/s

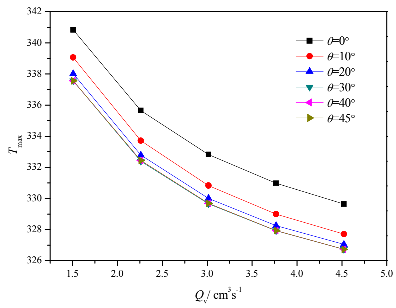
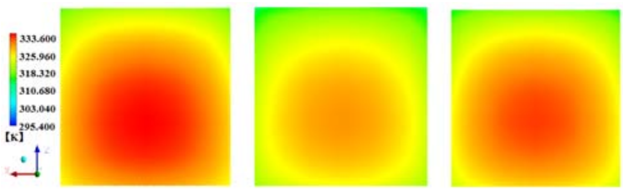


Fig. 12 T_{max} of heat source surface



(a) θ=0° (b) θ=30° (c) θ=45°

Fig. 13 Temperature distribution on the surface of chips

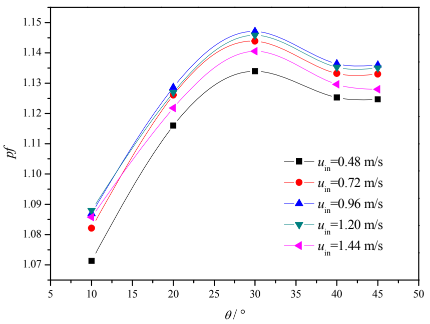


Fig. 14 Variation thermal performance factor with different pin fin located angle (model 2)

4. Summary & Conclusions

In this work, the pin fin porosity and rotated angle were numerically investigated for enhancing thermal performance of MMC heat sink. The main conclusions are as follows:

1. To increase the amount of pin fin or rotating pin fin can efficiently enhance the thermal performance of heat sink, but there exists the optimum values for both porosity and angle.

2. For the simulation cases, the optimum values of porosity and rotated angle are about 0.75 and 30^{o} respectively.

Acknowledgments

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